

Performance of a twin power piston low temperature differential Stirling engine powered by a solar simulator

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Abstract

This paper provides an experimental investigation on the performance of a low-temperature differential Stirling engine. In this study, a twin power piston, gamma-configuration, low-temperature differential Stirling engine is tested with non-pressurized air by using a solar simulator as a heat source. The engine testing is performed with four different simulated solar intensities. Variations of engine torque, shaft power and brake thermal efficiency with engine speed and engine performance at various heat inputs are presented. The Beale number, obtained from the testing of the engine, is also investigated. The results indicate that at the maximum simulated solar intensity of 7145 W/m^2 , or heat input of 261.9 J/s , with a heater temperature of 436 K , the engine produces a maximum torque of 0.352 N m at 23.8 rpm , a maximum shaft power of 1.69 W at 52.1 rpm , and a maximum brake thermal efficiency of 0.645% at 52.1 rpm , approximately.

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1. Introduction

The low temperature differential (LTD) Stirling engine is a type of Stirling engine that can operate with a low temperature heat source. There are many low temperature heat sources available including solar energy. The LTD Stirling engine then provides the possibility of direct conversion of solar energy to mechanical work.

A LTD Stirling engine can be run with small temperature difference between the hot and cold ends of the displacer cylinder (Rizzo, 1997). LTD Stirling engines provide value as demonstration units, but they immediately become of interest when considering the possibility of power generation from many low-temperature waste heat

sources in which the temperature is less than 100°C (Van Arsdell, 2001). Some characteristics of the LTD Stirling engine are as follow (Rizzo, 1997):

- (1) Displacer to power piston swept volumes ratio is large.
- (2) Diameters of displacer cylinder and displacer are large.
- (3) Displacer length is short.
- (4) Effective heat transfer surfaces on both end plates of the displacer cylinder are large.
- (5) Displacer stroke is small.
- (6) Dwell period at the end of the displacer stroke is rather longer than the normal Stirling engine.
- (7) Operating speed is low.

The LTD Stirling engine has been studied by many researchers. Many studies in the field of LTD Stirling

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Nomenclature

A	absorber area (m^2)	q_{in}	actual heat input to the engine (J/s)
C_p	specific heat of water at constant pressure (4186 J/kg K)	q_s	total heat input from heat source (J/s)
E_H	heat source efficiency	r	dynamometer brake drum radius (0.1465 m)
E_{BT}	Brake thermal efficiency	S	spring balance reading (N)
f	engine frequency (Hz)	T	engine torque (N m)
H	heating value of the gas used (J/kg)	T_C	cooler wall temperature (K)
I	average intensity on absorber plate (W/m^2)	T_H	heater wall temperature (K)
m_f	mass of gas burned (kg)	T_{w1}	initial water temperature (K)
m_w	mass of water to absorb heat (kg)	T_{w2}	maximum water temperature after the heat source has been turned off (K)
N	engine speed (rpm, rps)	t_1	initial time at water temperature of T_{w1} (s)
N_B	Beale number ($\text{W}/\text{bar cc Hz}$)	t_2	final time at water temperature of T_{w2} (s)
P	shaft power (W)	V_P	power-piston swept volume (cc)
p_m	engine mean pressure (bar)	W	loading weight (N)

engines have been investigated in the authors' former work (Kongtragool and Wongwises, 2003a). Some works related to the LTD Stirling engine are as follow (Kongtragool and Wongwises, 2003a).

Haneman (1975) studied the possibility of using air with low temperature sources. An unusual engine, in which the exhaust heat was still sufficiently hot to be useful for other purposes, was constructed. Spencer (1989) reported that, in practice, such an engine would produce only little useful work relative to the collector system size, and would give little gain compared to the additional maintenance required.

A simply constructed low-temperature heat engine modeled on the Stirling engine configurations was patented by White (1983). He suggested improving performance by pressurizing the displacer chamber. Efficiencies were claimed to be around 30%, but this can be regarded as quiet high for a low-temperature engine.

O'Hare (1984) patented a device passing cooled and heated streams of air through a heat exchanger for changing the pressure of air inside the bellows. The practical usefulness of this device was not shown in detail as in the case of Haneman's work.

Rizzo (1997) and Senft (1993) reported that Kolin experimented with 16 LTD Stirling engines, over a period of 12 years. Kolin presented a model that worked on a temperature difference between the hot and cold ends of the displacer cylinder as low as 15 °C.

After Kolin published his work, Senft (Van Arsdell, 2001) made an in-depth study of the Ringbom engine and its derivatives, including the LTD engine. Senft's research in LTD Stirling engines resulted in a most interesting engine, which had an ultra-low temperature difference of 0.5 °C. It has been very difficult to create any development better than this result. Senft's work (Senft, 1991), showed the motivation in the use of Stirling engine, their target was to develop an engine operating with a temperature difference of 2 °C or lower.

Senft (1993) described the design and testing of a small LTD Ringbom Stirling engine powered by a 60° conical reflector. He reported that the tested 60° conical reflector, producing a hot end temperature of 93 °C under running conditions, worked very well.

Iwamoto et al. (1997) compared the performance of a LTD Stirling engine powered by the hot spring heat with a high temperature differential Stirling engine powered by the combustion gas. Finally, they concluded that the LTD Stirling engine efficiency at its rated speed was approximately 50% of the Carnot efficiency. However, the swept volume ratio of their LTD Stirling engine was approximately equal to that of a conventional Stirling engine. Then its performance seemed to be the performance of a common Stirling engine operating at a low operating temperature.

Kongtragool and Wongwises (2003b) made a theoretical investigation on the Beale number for LTD Stirling engines. The existing Beale number data for various engine specifications from the literature were collected. They concluded that the Beale number for a LTD Stirling engine could be found from the mean-pressure power formula.

Kongtragool and Wongwises (2005a) investigated, theoretically, the power output of a gamma-configuration LTD Stirling engine. Former works on Stirling engine power output calculations were studied and discussed. They pointed out that the mean-pressure power formula was the most appropriate for LTD Stirling engine power output estimation. However, the hot-space and cold-space working fluid temperature was needed in the mean-pressure power formula.

Kongtragool and Wongwises (2005b) presented the optimum absorber temperature of a once-reflecting full-conical reflector for a LTD Stirling engine. A mathematical model for the overall efficiency of a solar-powered Stirling engine was developed. Limiting conditions of both

maximum possible engine efficiency and power output were studied. Results showed that the optimum absorber temperatures obtained from both conditions were not significantly different and the overall efficiency in the case of the maximum possible engine power output was very close to that of the real engine of 55% Carnot efficiency.

As described above, there has been only one paper published on a LTD Stirling engine powered by solar energy (Senft, 1993). However, this engine was a Ringbom Stirling engine. There is no published paper on the experimental results of a kinematic, twin power piston, LTD Stirling engine.

Compared with the high temperature differential (HTD) Stirling engine, the LTD Stirling engine is better in its ability to function effectively both in LTD and HTD ranges. On the contrary, the HTD Stirling engine can neither function effectively nor function at all in the LTD range. Moreover, the LTD Stirling engine also has lower engine speed. Consequently, its engine parts are smaller and vibrate less. Therefore, specially strong materials are not needed for making engine parts. Thus, it is interesting to further study the LTD Stirling engine in details.

In this paper, the testing performance of a LTD Stirling engine is presented. The LTD Stirling engine tested in this paper is a kinematic, single-acting, twin power piston, gamma-configuration, LTD Stirling engine using non-preserved air as a working fluid. The heat source used is a solar simulator made from a tungsten halogen lamp.

The twin-power piston engine developed is aimed to be used with a solar concentrator in the next step of our future work. This engine is good in that it yields as much power as a two-cylinder single-acting engine or a single-cylinder double-acting engine. However, both the two-cylinder single-acting engine and the single-cylinder double-acting engine have to use two displacer cylinders. Hence, it is very difficult or even impossible to use those engines with a conventional solar concentrator. Therefore, the twin-power piston engine is more likely to be developed into a compact engine with high power piston swept volume that can be used with a conventional solar concentrator.

The twin-power piston engine developed is specifically designed to have two power cylinders directly installed on a cooler plate. Thus, it is not necessary to use transfer ports, which results in as minimal dead volume as possible. Making a cooler plate part of a cooler not only helps in reducing number of parts, but also helps in ventilating heat at the power cylinders. Furthermore, the displacer is also designed to serve as a regenerator. As a result, the engine structure is simple and uses as minimal part as possible.

This engine design is based on a principle of multifunctional capability of parts. As a result, it requires lower production cost and has simpler engine structure. Since the LTD Stirling engine can work in low temperature, it is possible to use simple solar concentrator such as axicon or conical reflector whose structure is easier to construct than the parabolic dish concentrator. Therefore, the production cost of this part is also lower.

2. Experimental apparatus

Details of experimental apparatus are as follows:

2.1. Test engine

The schematic diagram of the test engine is shown in Fig. 1. Main engine design parameters are listed in Table 1. This engine is a laboratory-scale LTD Stirling engine. It is designed in single-acting, gamma-configuration with a twin power piston. Since the gamma configuration provides a large regenerator heat transfer area (Iwamoto et al., 1997) and is easy to be constructed (Rizzo, 1997), this configuration is used in the present study. Two power cylinders are directly connected to the cooler plate to minimize the cold-space and transfer-port dead volume. The cooling water pan is a part of the cooler plate.

In order to make the engine compact and to minimize the number of engine parts, a simple crank mechanism is used in this engine. The crankshaft is made from a steel shaft, two crank discs and a crank pin. The crank pin is connected to the displacer connecting rod. The crankshaft is supported by two ball bearings. Two steel flywheels are attached to both ends of the crankshaft. These flywheels also act as the crank discs for the power piston.

The power cylinders are made from a steel pipe. The cylinder bores are finished by turning. The power pistons are made from a steel pipe and plate. The power piston cylindrical surfaces are brass lined and then turned to match the power cylinder bores. The clearance between piston and bore is 0.02 mm, approximately. Oil grooves, 1 mm × 1 mm with 10 mm spacing are provided on the cylindrical piston surfaces.

The displacer cylinder and displacer cylinder head is made from a 0.5 mm thick stainless steel Indian pan. The displacer swept volume is 6394 cc with 32 cm bore and 7.95 cm stroke. The power piston bore and stroke is 8.3 cm and 8.26 cm, respectively. The swept volume ratio of this engine is 7.15.

The displacer of this engine also serves as a regenerator. This moving regenerator is made from a perforated steel sheet with a round-hole pattern. The stainless steel pot scourer is used as a regenerator matrix. The clearance between the displacer and its cylinder is 1 mm, approximately.

A stainless steel pipe is used as the displacer rod. The displacer rod is guided by two brass bushings placed in the displacer rod guide housing. Two rubber seals are used to prevent leakage through the displacer rod guide bushings.

The power piston and displacer connecting rod are made from a steel rod. Both ends of the connecting rod are attached to two ball bearings housings.

2.2. Testing facilities

The schematic diagrams of the experimental apparatus are shown in Figs. 2–4. The photograph of engine with test-

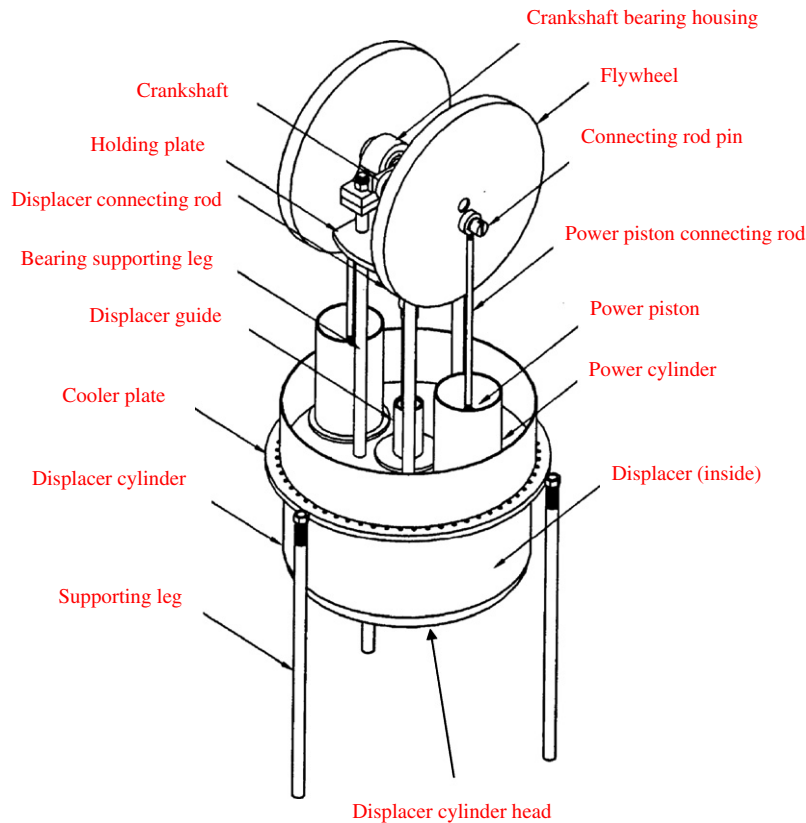


Fig. 1. Schematic diagram of the tested Stirling engine.

Table 1
Main engine design parameters

Mechanical configuration		Gamma
Power piston	Bore $\times$ stroke (cm)	8.3 $\times$ 8.26
	Swept volume (cc)	893.8
Displacer	Bore $\times$ stroke (cm)	32 $\times$ 7.95
	Swept volume (cc)	6393.8
Swept volume ratio		7.15
Phase angle		90°

ing facilities is shown in Fig. 5. Details of the testing facilities are as follows:

Since the engine speed is low, a rope-brake dynamometer can be used to measure the engine torque. A flywheel, 29.3 cm in diameter, is used as a brake drum. The braking load is measured by the loading weight and the spring balance reading. A photo tachometer with ± 0.1 rpm accuracy is used to measure the engine speed.

The cooling water is supplied from a cooling water system. The cooler wall temperature, T_C , is measured by three T-type thermocouples. The heater wall temperature, T_H , is

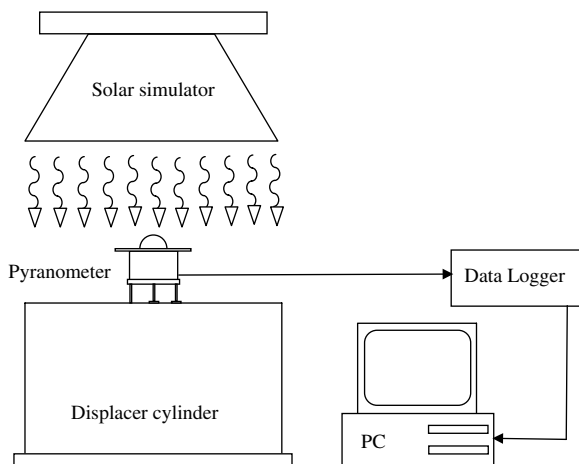


Fig. 2. Schematic diagram of the simulated solar intensity test.

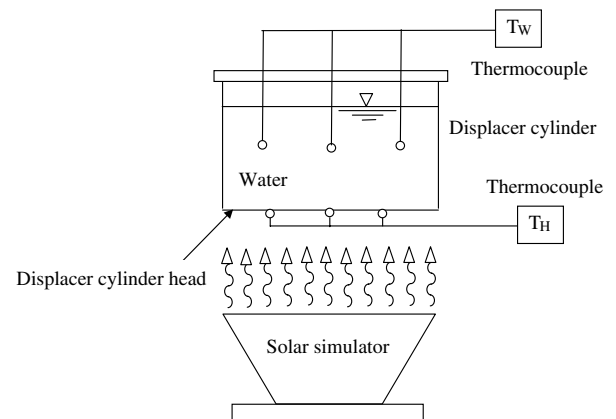


Fig. 3. Schematic diagram of the heat source efficiency test.

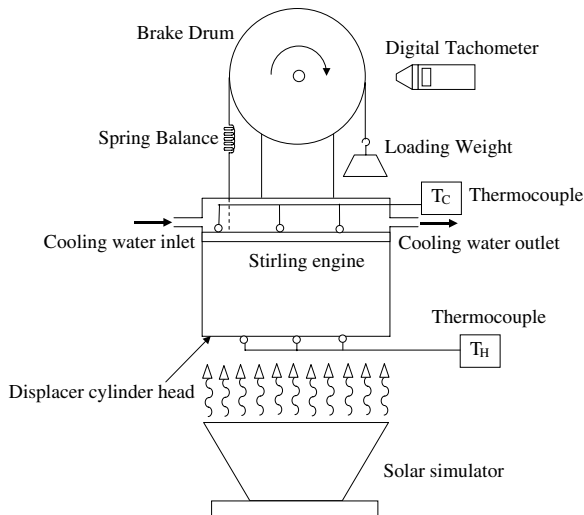


Fig. 4. Schematic diagram of performance test for the twin power piston engine powered by solar simulator.

measured by three K-type thermocouples. These thermocouples are mounted at three positions, equally spaced around the plate (120° apart) at 15 cm from the center.

A solar simulator consisting of a 1 kW tungsten halogen lamp (Osram Haloline 64740L J R7s) is used to power the engine. The intensity on the absorber plate or displacer cyl-

inder head is measured by a pyranometer (Lambert model 00.16103.000000 CM 3).

A data logger (DataTaker model DT 50) and a personal computer are used to collect data from the thermocouples and pyranometer. The accuracy of the temperature measurement is $\pm 0.1^\circ\text{C}$. The sensitivity of the intensity measurement obtained from the pyranometer is $\pm 0.05\%$. The calibrated pyranometer constant is $23.66 \mu\text{V}/(\text{W}/\text{m}^2)$.

3. Experimental procedures

3.1. Intensity test

The actual intensity on the absorber plate is needed for engine performance calculation. The experiment for determination of the actual intensity on the engine absorber at various distances from halogen lamp to absorber, has been carried out first. A pyranometer is used to measure the intensity on the displacer cylinder head that acts as the absorber plate. A data logger and a personal computer are used to collect data from the pyranometer. The testing procedure is as follow:

The displacer cylinder and the halogen lamp are put on the stand. The distance from the halogen lamp to the absorber plate is set as required. The pyranometer is placed on the absorber plate at seven positions. One position is at

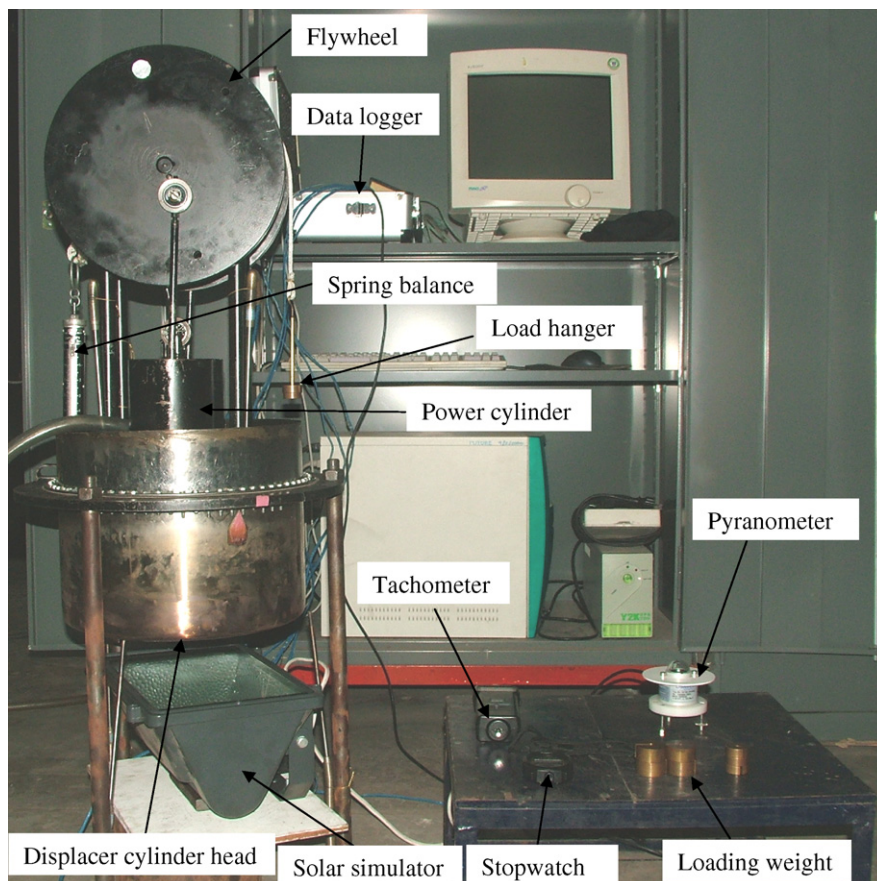


Fig. 5. Engine with testing facilities.

the center of the absorber. The other six positions are at the middle of the radius and 60° apart. The pyranometer is connected to the data taker and computer. Turn on the halogen lamp and collect the intensity at that position and then place the pyranometer to other positions.

The testing can be repeated for other intensities by changing the distance between lamp and absorber. The test results from those twelve distances are shown in Fig. 6.

3.2. Heat source test

The actual or useful heat input can not be determined directly while the engine is running because of the difficulties caused by instrumentation. Therefore, this experiment has been carried out before the performance test, to determine the actual heat input to the engine. The testing procedure is as follow:

The displacer cylinder insulated with 25.4 mm thick insulator is put on the stand. Five kilogram of water is put into the displacer cylinder. This water is used to absorb heat from the solar simulator. The absorbed heat is the useful heat input to the displacer cylinder.

The thermocouples for measuring the displacer cylinder head wall temperature and the water temperature are installed. Three T-type thermocouples are used to measure

the water temperature in the displacer cylinder. While three K-type thermocouples are used to measure the displacer cylinder head wall temperatures.

The halogen lamp is placed at the required distance, underneath the displacer cylinder head. The initial water temperature is registered and the halogen lamp is turn on. Before the boiling point is reached, all temperatures are taken every 1 min interval using a data logger and a personal computer.

The testing is repeated with another heat input by changing the distance between the halogen lamp and the absorber. The test results from four intensities are shown in Table 2.

3.3. Performance test

Before the engine is started, all thermocouples are connected to the data logger and computer. The cooling water system is connected to the engine cooling pan. The cooling water flow rate is adjusted in order to keep the water level in the cooling pan constant.

The halogen lamp is placed underneath the displacer cylinder head (or absorber plate) at a given distance. Some lubricating oil is ejected into the power pistons, cylinders, and the displacer guide bushing. The halogen lamp switch

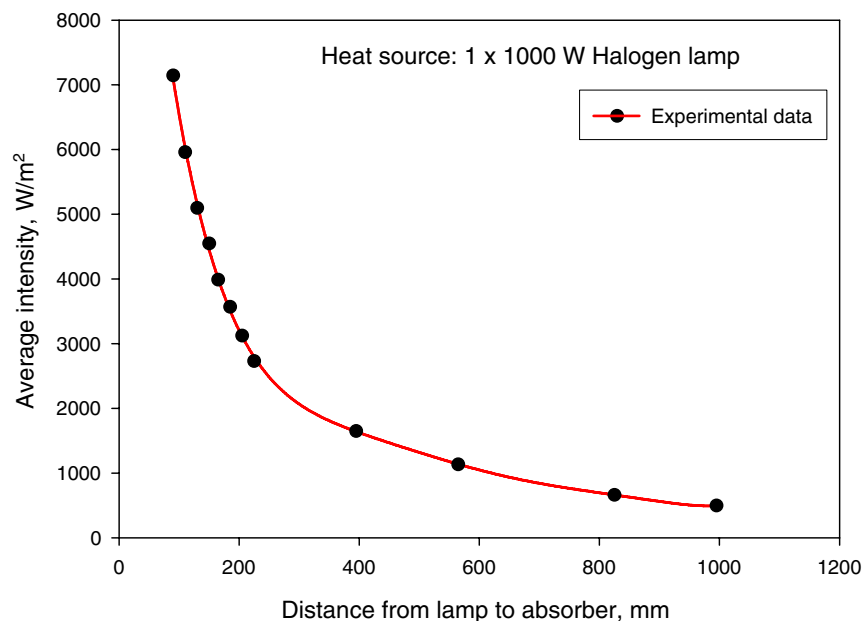


Fig. 6. Average intensity on absorber plate versus distance from lamp to absorber.

Table 2

Maximum engine performance and Beale number at $T_C = 307$ K

I (W/m ²)	E_H (%)	q_{in} (J/s)	T_H (K)	τ	T (N m)	P_{max} (W)	E_{BT} (%)	N_B (10 ⁻³ W/bar cc Hz)
4549	71.36	220.3	399	0.7694	0.236 @ 21.3 rpm	0.879 @ 44.3 rpm	0.399 @ 44.3 rpm	1.3319
5097	61.26	230.2	409	0.7506	0.260 @ 21.6 rpm	0.969 @ 46.5 rpm	0.421 @ 46.5 rpm	1.3988
5961	56.03	242.1	419	0.7327	0.287 @ 23.9 rpm	1.22 @ 47.3 rpm	0.504 @ 47.3 rpm	1.7314
7145	52.84	261.9	436	0.7041	0.352 @ 23.8 rpm	1.69 @ 52.1 rpm	0.645 @ 52.1 rpm	2.1774

is then turned on. The displacer cylinder head is heated up until it reaches the operating temperature. The engine is then started and run until the steady condition is reached.

The engine is loaded by adding a weight to the dynamometer. After that, the engine speed reading, spring balance reading and all temperatures from the thermocouples are collected. Another loading weight is added to the dynamometer until the engine is stopped.

Testing can be repeated with another simulated intensity by changing the distance from the lamp to absorber.

In this study, engine tests are performed by using four distances from lamp to absorber. The average simulated

intensities on the absorber plate are 4549, 5097, 5961, and 7145 W/m².

4. Experimental results and discussion

Figs. 7–15 show the engine test results. In these figures, the engine torque is calculated from

$$T = (S - W)r \quad (1)$$

where S is the spring balance reading, W the loading weight, and r is the brake drum radius. The actual shaft power can be calculated from

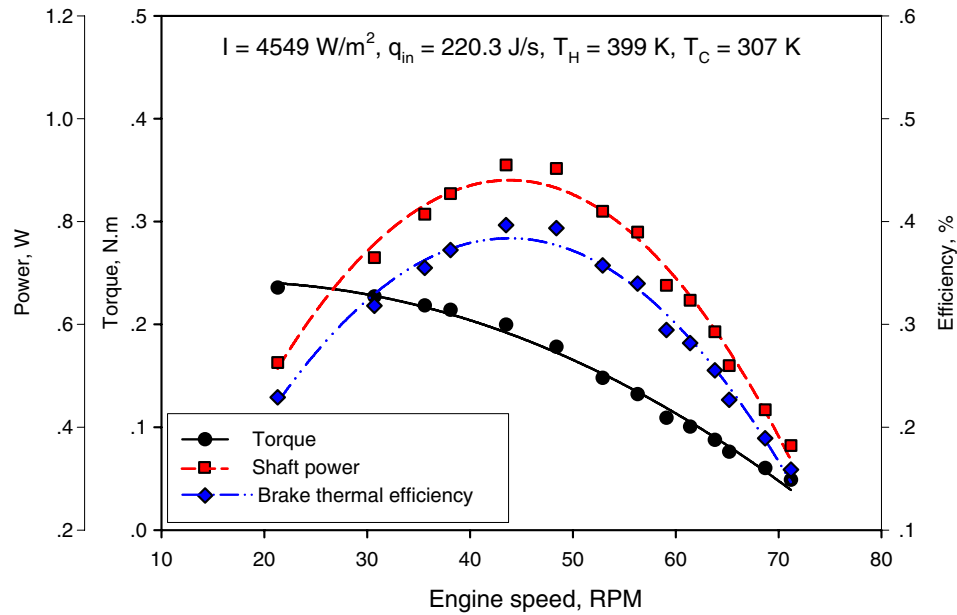


Fig. 7. Engine performance at simulated solar intensity of 4549 W/m² (or 220.3 J/s heat input).

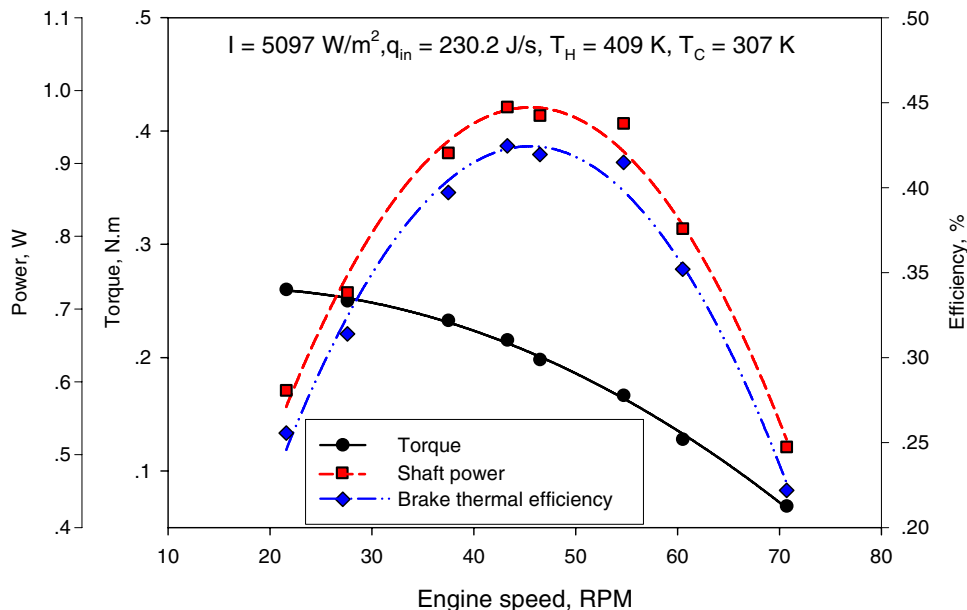


Fig. 8. Engine performance at simulated solar intensity of 5097 W/m² (or 230.2 J/s heat input).

$$P = 2\pi TN \quad (2)$$

where N is the engine speed. E_{BT} is the brake thermal efficiency calculated from

$$E_{BT} = P/q_{in} \quad (3)$$

where q_{in} is the actual heat input to the engine. The actual heat input to the engine can be determined from

$$q_{in} = E_H q_S \quad (4)$$

where E_H is the heat source efficiency and q_S is the total heat input from the heat source.

For a gas burner, the heat source (burner) efficiency can be calculated from European Standard (EN 30 Edition 2, 1979) as follows:

$$E_H = \frac{m_w C_p (T_{w2} - T_{w1})}{m_f H} \quad (5)$$

where m_w is the mass of water to absorb heat, C_p the specific heat of water at constant pressure, T_{w1} and T_{w2} are, respectively, the initial water temperature and the maximum water temperature after the burner has been turned off, m_f the mass of gas burned, and H is the heating value

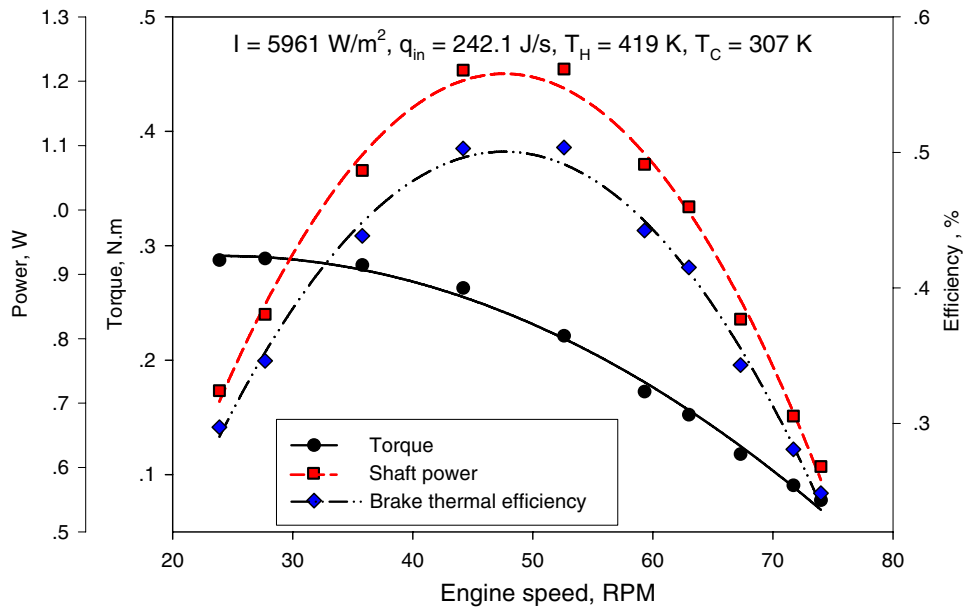


Fig. 9. Engine performance at simulated solar intensity of 5961 W/m^2 (or 242.1 J/s heat input).

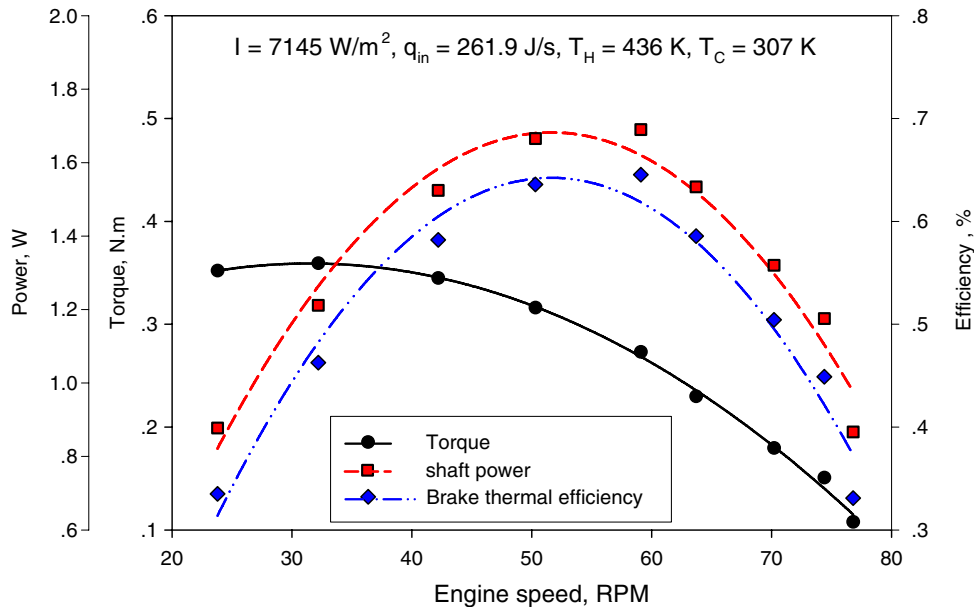


Fig. 10. Engine performance at simulated solar intensity of 7145 W/m^2 (or 261.9 J/s heat input).

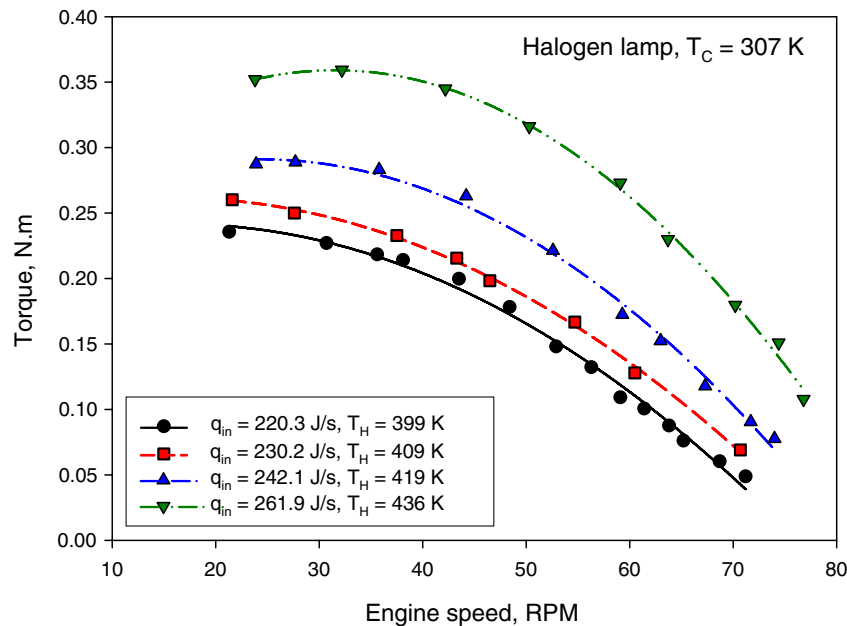


Fig. 11. Variations of engine torque at various simulated solar intensities.

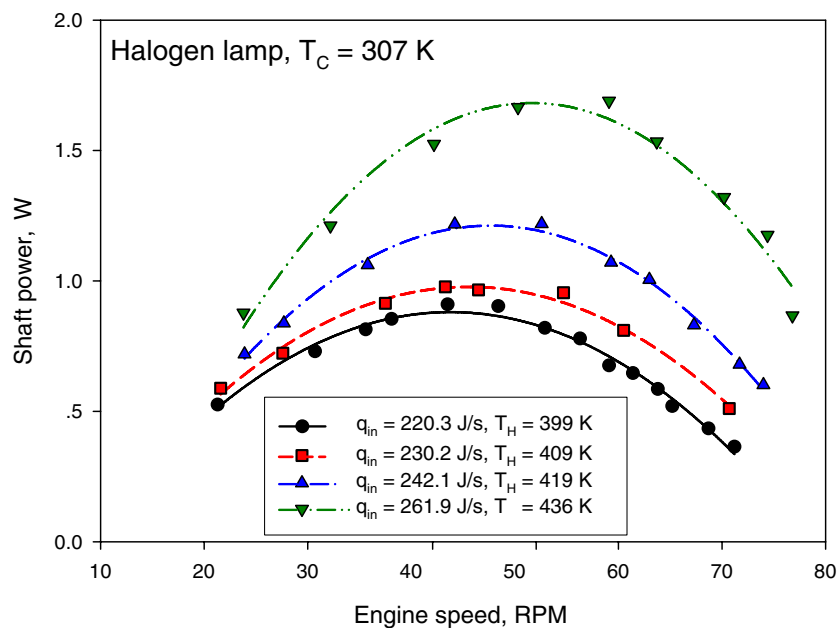


Fig. 12. Variations of engine shaft power at various simulated solar intensities.

of the gas. For an absorber having area A , with an intensity from a solar simulator of I , the heat source efficiency can also be determined from Eq. (5) by replacing the product of $m_f H$ with $IA(t_2 - t_1)$, where t_1 is initial time at water temperature of T_{w1} and t_2 is final time at water temperature of T_{w2} .

The actual heat input to the engine, q_{in} , at the average simulated intensities of 4549, 5097, 5961, and 7145 W/m² was experimentally determined by using water to absorb this heat. The actual heat input to the engine, the heat source efficiency and the absorber temperatures resulting from these simulated intensities are shown in Table 2.

The engine performances at average simulated intensities of 4549, 5097, 5961, and 7145 W/m² are shown in Figs. 7–10, respectively. The performance curves of this engine are described as follow.

In the engine test, after the load is gradually applied to the engine, the engine speed will be gradually reduced. At a certain amount of load applied, the engine will stop. The characteristics are represented by the curve of torque versus engine speed. In these figures, it can be noted that the engine torque decreases with increasing engine speed.

The variations of shaft power with engine speed are also shown in these figures. The shaft power increases with

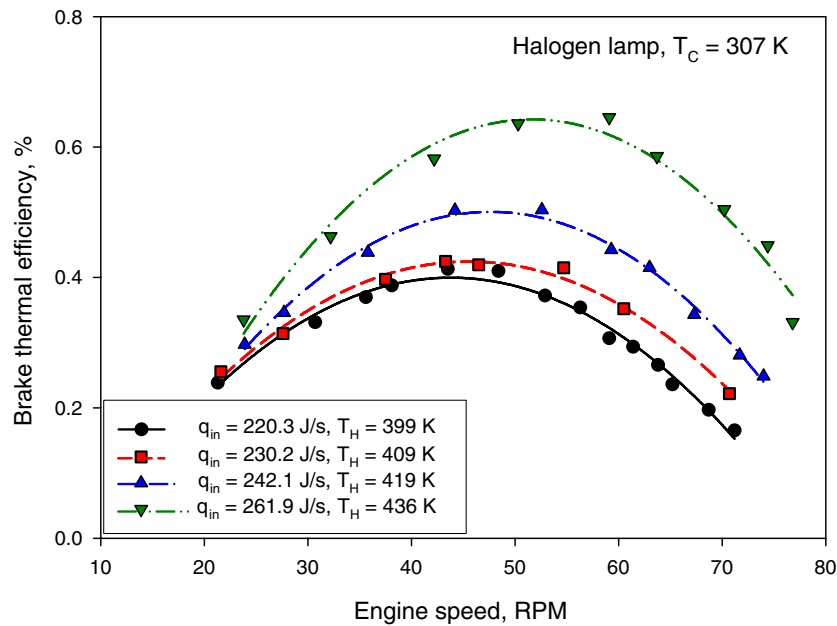


Fig. 13. Variations of brake thermal efficiency at various simulated solar intensities.

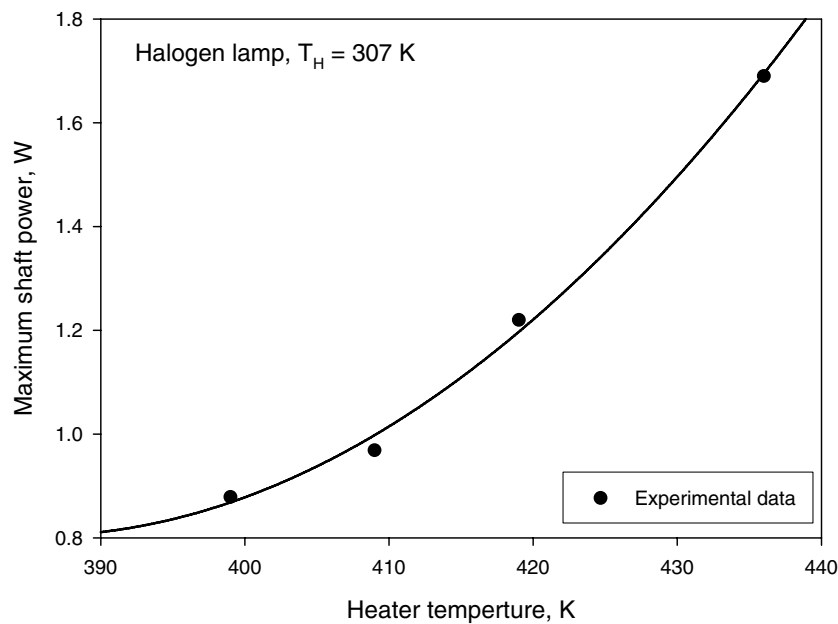


Fig. 14. Variations of maximum shaft power with heater temperature.

increasing engine speed until the maximum shaft power is reached and then decreases with increasing engine speed. The decreasing in shaft power after the maximum point, results from the friction that increases with increasing speed together with inadequate heat transfers at higher speed.

Since the brake thermal efficiency is the shaft power divided by a constant heat input, the curve of brake thermal efficiency has the same trend as the shaft power. The main technical problem is the engine gives slightly low thermal efficiency. This may be caused by engine parts machining problems encountered during manufacturing of this engine.

This causes the engine to have some misalignments and higher friction. Another cause is that the engine operates at a relatively low temperature. The heat source efficiency, distance from lamp to displacer head, the displacer head thickness, and convection heat loss also affect the thermal efficiency.

Performance improvement in terms of design and construction can be achieved in many ways. For example, alignment and precision of engine parts can be improved by using standard parts (e.g. using rod ends, instead of connecting rod big end and small end made from ball bearings) and professional technicians who have specific experiences

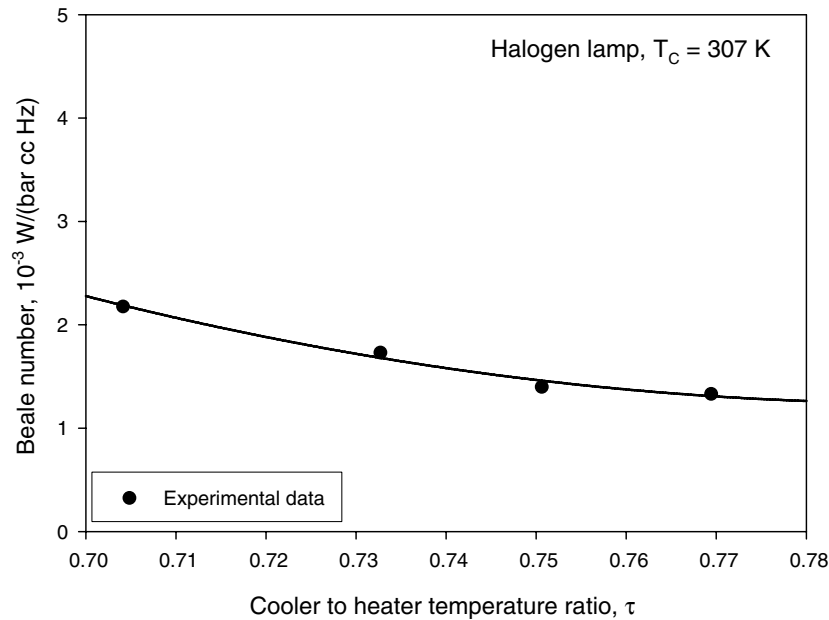


Fig. 15. Variations of Beale number with temperature ratio.

in constructing or rebuilding engines. Friction can be reduced by changing seal used at the displacer rod from rubber seal to oil grooves seal, as used in the seal of power piston. Moreover, flywheel weight can be reduced by decreasing weight of power piston, which can be done by making piston skirt and piston head thinner and strengthening them with reinforced stiffeners. The displacer weight can be reduced by changing a regenerator matrix from stainless steel to aluminum.

After improving the engine as mentioned above, the performance of the engine could be increased to an acceptable level.

Figs. 11–13 show the variations of the engine torque, shaft power, and brake thermal efficiency at various heat inputs with engine speed. As expected, the greater engine performance results from the higher heat input. An increase of the engine torque, shaft power and brake thermal efficiency is shown to also depend on the heater wall temperature. The values of the maximum torque, shaft power and brake thermal efficiency are summarized in Table 2.

The maximum shaft power at various heat inputs is plotted against the heater wall temperature in Table 2. The plot is shown in Fig. 14. In this figure, the shaft power increases as a function of the heater wall temperature. Therefore, the greater maximum shaft power will be obtained from the higher heater wall temperature.

It is clearly seen that this engine could function by using heater wall temperature ranging between 126 and 163 °C only, with concentrated intensity of 4549–7145 W/m². If the direct solar intensity is 120 W/m², the concentration factor value will fall roughly between 38 and 60. An easily constructed and inexpensive solar concentrator such as axicon or conical reflector could simply yield a concentration factor value and temperature in this range. To achieve

a desired high concentration factor value, the principle of two-stage concentration can also be applied.

Table 2 also shows the Beale number obtained from this engine at various heat inputs. These Beale numbers are plotted against the ratio of the temperature, $\tau = T_c/T_H$. The plot is shown in Fig. 15. The Beale number is calculated from the Beale formula (Kongtragool and Wongwises, 2003b, 2005a)

$$N_B = P/(p_m V_P f) \quad (6)$$

where p_m is engine mean pressure in bar, V_P the power-piston swept volume (cc), and f is the engine frequency (Hz). For a non-pressurized engine, $p_m = 1$ bar is used in the calculation, as described by Senft (1993). From this figure, it can be seen that the Beale number decreases with increasing temperature ratio. In other words, it can be said that the Beale number increases with increasing heater wall temperature.

5. Conclusions

Results from this study indicate that the engine performance increases with increasing simulated solar intensity. The engine torque, shaft power, brake thermal efficiency, speed, and heater wall temperature also increase with increasing simulated intensity. In fact, it can be said that the maximum engine torque, shaft power, and brake thermal efficiency increase with increasing heater wall temperature. The Beale number of this engine increases with decreasing temperature ratio or with increasing heater wall temperature.

At the maximum simulated solar intensity of 7145 W/m², or heat input of 261.9 J/s, the engine produced a maximum torque of 0.352 N m at 23.8 rpm, a maximum shaft power

of 1.69 W at 52.1 rpm, and a maximum brake thermal efficiency of 0.645% at 52.1 rpm, approximately, with a heater temperature of 436 K.

Even though these figures are not so high, if we realize that the solar-powered Stirling engine is powered by a free heat source, this study would probably be more worthwhile as a starting point for the research of cheap solar-powered engines manufactured for rural and remote areas. The engine performance can be improved by increasing the precision of the engine parts and the heat source efficiency. The engine performance could be further increased if a better working fluid, e.g. helium or hydrogen, is used, instead of the air and/or by operating the engine at some degree of pressurization.

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